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A FINAL REPORT ON THE DESIGN OF A COOLING SYSTEM FOR THE ADVANCED TURBINE AEROTHERMAL RESEARCH RIG (ATARR)

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This document will serve as the final report for the ATARR turbine cooling system design effort which has been conducted at Calspan ATC. The material provided herein presents the cooling system requirements which have driven the design, and provides examples of systems used at other facilities to meet similar requirements. Two specific cooling options are described and a synopsis is given of the efforts that have led to the suggested design. Sufficient information is provided for the reader to elect either system and proceed with final design considerations and eventual construction.

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NOMENCLATURE

A	area
a	acceleration
C_d	flow coefficient
C_p	specific heat
D	diameter
d	distance
dx	differential length
E	energy
F	force
H	total enthalpy
h	heat-transfer coefficient or enthalpy per unit mass
L	length
m	mass
N	number of tubes
OD	outer diameter
P	pressure
Q	heat energy
q	heat rate per unit time
R	gas constant
S	tensile strength
T	temperature

t	time or thickness
V	volume
v	velocity
w	wall thickness
wt	weight
γ	ratio of specific heats
λ	length constant
ρ	density
θ	nondimensional temperature
τ	time constant

Superscripts

*	nondimensional quantity
·	flow rate, per unit time

Subscripts

c	cold flow, cooling gas
cf	core flow
g	gas
h	hot flow
i	initial
m	mixed flow, metal
r	reservoir
t	throat
w	wall

1.0 - INTRODUCTION

This document will serve as the final report for the ATARR turbine cooling system design effort which has been conducted at Calspan ATC. Unfortunately, due to time constraints the design effort was not completed to the degree that we would have liked. The material provided herein presents the cooling system requirements which have driven the design, and provides examples of systems used at other facilities to meet similar requirements. Two specific cooling options are described and a synopsis is given of the efforts that have led to the suggested design. Sufficient information is provided for the reader to elect either system and proceed with final design considerations and eventual construction.

1.1 - System Requirements

The primary requirement of the turbine cooling system is to match the core flow to cooling gas ratios of mass flow rate, temperature and pressure. The system must be capable of matching a relatively large range of these ratios to accommodate the full spectrum of turbines which might be tested in the ATARR. The matching of the desired ratios is complicated by the fact that in blow down facilities such as ATARR, the core flow temperature and pressure change with time as the supply tank empties. Therefore, the cooling gas properties must follow the same function in time that the core flow gas properties do in order to maintain the correct ratios throughout the test period.

The changing core flow gas properties in the ATARR test section can be modeled by an isentropic blowdown process. This modeling method for the blow down facility was

described by Epstein [1985] in the design of the MIT blowdown turbine facility and employed by Haldeman [1991] in the estimation of the mass flow rate out of the ATARR supply tank. Assuming an ideal gas, the time dependent equations for the core flow properties can be written as given below.

$$\dot{m}_{cf}(t) = \dot{m}_{cf}(0)\left[1+\frac{t}{\tau}\right]^{\frac{-1-\gamma}{\gamma-1}} \tag{1}$$

$$P_{cf}(t) = P_{cf}(0)\left[1+\frac{t}{\tau}\right]^{\frac{-2\gamma}{\gamma-1}} \tag{2}$$

$$T_{cf}(t) = T_{cf}(0)\left[1+\frac{t}{\tau}\right]^{-2} \tag{3}$$

These isentropic equations were used throughout the cooling system design to model the core flow properties.

It would be very difficult for a cooling system to exactly match the core flow property ratios over the entire test period. Therefore, it is crucial that the cooling system have provisions for accurate measurement of the actual cooling gas properties that are delivered to the turbine. The accuracy of these measurements must meet the overall ATARR goal of being able to measure turbine efficiency to $\pm$ 0.25% of the true value within a 95% confidence interval.

If the ratios of interest were to be matched exactly, then the cooling system could become extremely complex and costly. It is felt that such a system is not practical. Therefore, it is necessary to emphasize two remaining requirements: the system should be economic to develop, and it should be easy to operate and maintain. The development costs associated with a complex system are often difficult to predict and great foresight is required early in the design process. The operation and maintenance of the cooling system will dictate its success for years to come. Additional time and money are well spent in consideration of these requirements during the design of the system.

1.2 - Background

Prior to committing to a particular scheme, it is prudent to research what approach others may have taken in solving similar design problems. For short-duration turbine test facility cooling systems one is limited to Calspan ATC, MIT Gas Turbine Laboratory, (GTL), Oxford University, and the Von Karman Institute for Fluid Mechanics. All of these laboratories have addressed the difficult problem of injecting cooling gas into the core flow. Both Calspan and MIT have employed blowdown type cooling systems to deliver the cooling gas. The MIT system uses a refrigerated supply tank and a fast acting valve to supply cooling gas in their facility which operates for a test period on the order of 300 milliseconds. Calspan uses a very similar blowdown type cooling system. This system is very simple requiring only that the fast acting valve operate at the appropriate time. The more difficult problem is the one of measuring accurately the temperature and pressure at various stages of the flow system.

In the very early stages, a blowdown type cooling system was considered for the ATARR facility but was abandoned for a more sophisticated system which would theoretically provide the desired core flow to cooling gas ratios throughout the duration of the test period. This system was designed to use high frequency response, dynamic control valves to meter the flow through a heat exchanger. The system is still a viable one and because it is, will be discussed in greater detail later. Recently, after reviewing many of the complexities associated with the heat exchanger system a decision was made to revisit the simpler blowdown type system in order to provide an acceptable cooling system within the budget and time constraints of the ATARR. This decision lead to the current status of the design choices which will be described in the following sections.

2.0 - SYSTEM OPTIONS

The two turbine cooling systems which have been considered for the ATARR application are discussed in this report. First, the control valve/heat exchanger system is described followed by a description of the simpler blowdown system. A computer model that has been developed to help with the detailed design of the blowdown system will also be described. Some work remains to be done with this computer model, but, in principle, it appears to work satisfactorily.

2.1 - Control Cooling System

The control type cooling system was presented to the ATARR personnel by Dr. Kim in July of 1991. A report detailing the design was made available at that time. The version presented here is virtually unchanged from the original design. This is the system that we have referred to earlier as a more sophisticated system. The only major change is the replacement of the mylar diaphragm/air knife with a fast acting valve to avoid the problem of removing mylar fragments from the downstream catcher after every run.

2.1.1 - Control System Operation

A schematic of the proposed control-type cooling system is given in Figure 1. The gas is metered using a Fox control valve (or Venturi meter), then passed through a previously cooled heat exchanger (tube bundle) where the gas temperature is lowered to a value somewhat lower than the desired coolant gas temperature. Gas that has bypassed the

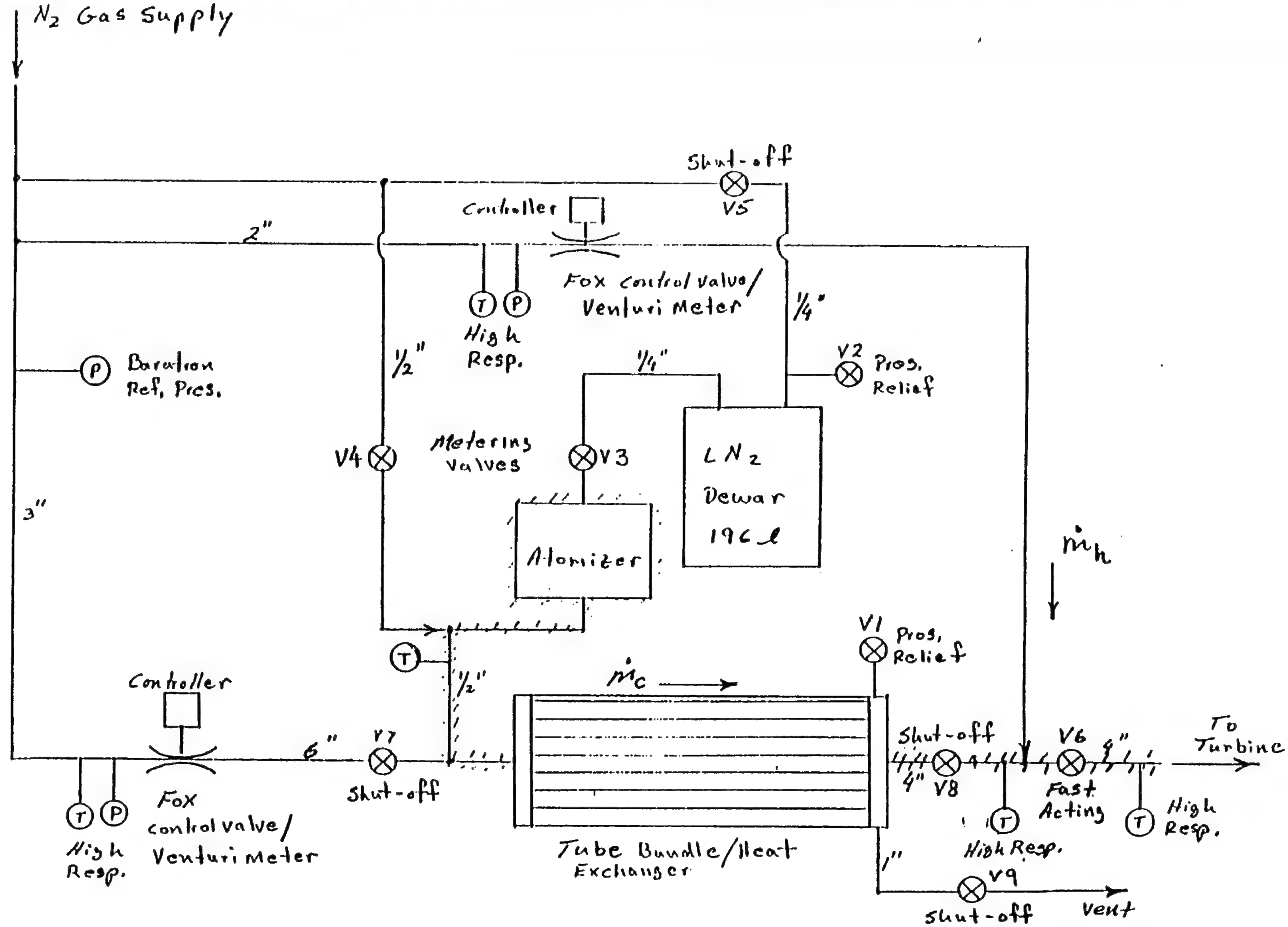

Figure 1 - Control Cooling System Schematic

heat exchanger via a separate line (the "hot" gas) is also metered, then mixed with the cold gas to obtain an exit gas flow at the required temperature. Cooling of the heat exchanger is accomplished by injecting cold N_2 gas obtained by mixing liquid nitrogen (LN_2) and N_2 gas.

The planned test procedure is as follows. Valves V7 and V8 are closed to isolate the heat exchanger from the turbine stage. The LN_2 tank is pressurized using N_2 gas from a supply capable of providing N_2 at the desired pressure. Valve V5 is opened and valves V3 and V4 are adjusted to obtain a cold gas flow at the desired temperature. The heat exchanger is cooled using this gas. Once the heat exchanger has been cooled to the desired temperature, valves V3, V4, and V5 are closed. Valves V7 and V8 are then opened, allowing the residual gas in the lines to vent through the cooling lines to the turbine stage (which is near vacuum), pre-cooling the turbine hardware and piping. The test is initiated by computer activation of the fast acting valve V6. The Fox venturi metering valves are controlled during the test such that the exit gas temperature and mass flow rate vary according to the desired rate. The coolant flow is terminated after the main valve is closed by closing valve V6.

2.1.2 - Control System Analysis

2.1.2.1 Control Valve Area Analysis

The two constraints that the coolant gas being supplied to the turbine stage must satisfy are: 1). The coolant gas temperature must decay at the same rate as the core flow gas temperature and 2). The coolant gas mass flow rate must decay at the same rate as that of the core flow mass flow rate. The first constraint can be satisfied by varying the ratio of

the hot to cold gas flow rate. The second constraint can be satisfied by varying the sum of the hot and cold gas control valve areas. Consider the system shown on Figure 2.

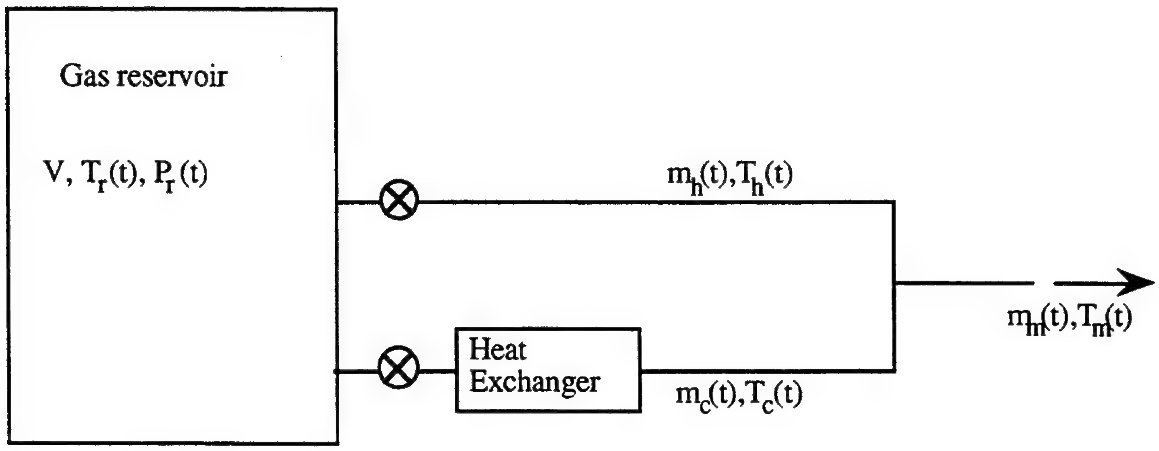

Figure 2 - Schematic of cooling system

The conservation equations dictate that

conservation of mass: $$\dot{m}_h(t) + \dot{m}_c(t) = \dot{m}_m(t) \tag{4}$$

conservation of energy:

$$\dot{m}_h(t)C_{p,h}T_h(t) + \dot{m}_c(t)C_{p,c}T_c(t) = \dot{m}_m(t)C_{p,m}T_m(t) \tag{5}$$

Solving the above equations, assuming $C_{p,h} = C_{p,c} = C_{p,m}$, yields the mass flow rates of the cold to hot gas as

$$\dot{m}_c(t) = \frac{T_h(t) - T_m(t)}{T_h(t) - T_c(t)}\dot{m}_m(t) \text{ and } \dot{m}_h(t) = \frac{T_m(t) - T_c(t)}{T_h(t) - T_c(t)}\dot{m}_m(t). \tag{6}$$

At any instant in time the mass flow rate and temperature of the coolant gas, $\dot{m}_m(t)$ and $T_m(t)$, are specified from the test requirements, and the variation of the hot and cold gas temperatures with time, $T_h(t)$ and $T_c(t)$, can be measured. The mass flow rates of the hot and cold gas streams can therefore be obtained at any instant from equations (6).

The mass flow rate through a control valve is given by

$$\dot{m} = C_d \rho_t A_t v_t \tag{7}$$

where the subscript t stands for the conditions at the throat. For choked flow, it can be shown that

$$A_t = \frac{\dot{m}\sqrt{RT_t}}{C_d\sqrt{\gamma}\,P_t} \tag{8}$$

where

$$P_t = P_r\left(1 + \frac{\gamma - 1}{2}\right)^{\frac{-\gamma}{\gamma - 1}} \tag{9}$$

$$T_t = T_r\left(1 + \frac{\gamma - 1}{2}\right)^{-1} \tag{10}$$

The throat area required to produce the desired mass flow rate can be obtained for any gas from the pressure and temperature of the gas upstream of the throat (P_r and T_r), and the C_d of the valve. For the venturi type control valves to be used in this cooling scheme, C_d~1.

A computer is be used to monitor and control the cooling system. Measurements of the temperatures and pressures in the hot and cold lines are used to calculate the appropriate control valve flow areas required to produce the desired coolant flow. A signal is then sent to the control valves to produce these choke areas. The advantage of having computer control is that it offers a way of producing the desired coolant flow conditions without the need for careful control of heat exchanger temperature and N_2 gas temperature and pressure.

There are many plug-in boards available with data transfer rates much higher than those required for this application. The A/D board manufacturer claims that a data transfer time of 0.1 ms can be expected. Preliminary calculations indicate that one cycle in this process should take no longer than ~15 ms, meaning that the venturi valve positions are

updated roughly 67 times per second. This would be more than adequate. Even if the cycle were to require 50 ms, the venturi position would still be updated 20 times per second. A schematic of the control system is shown on Figure 3.

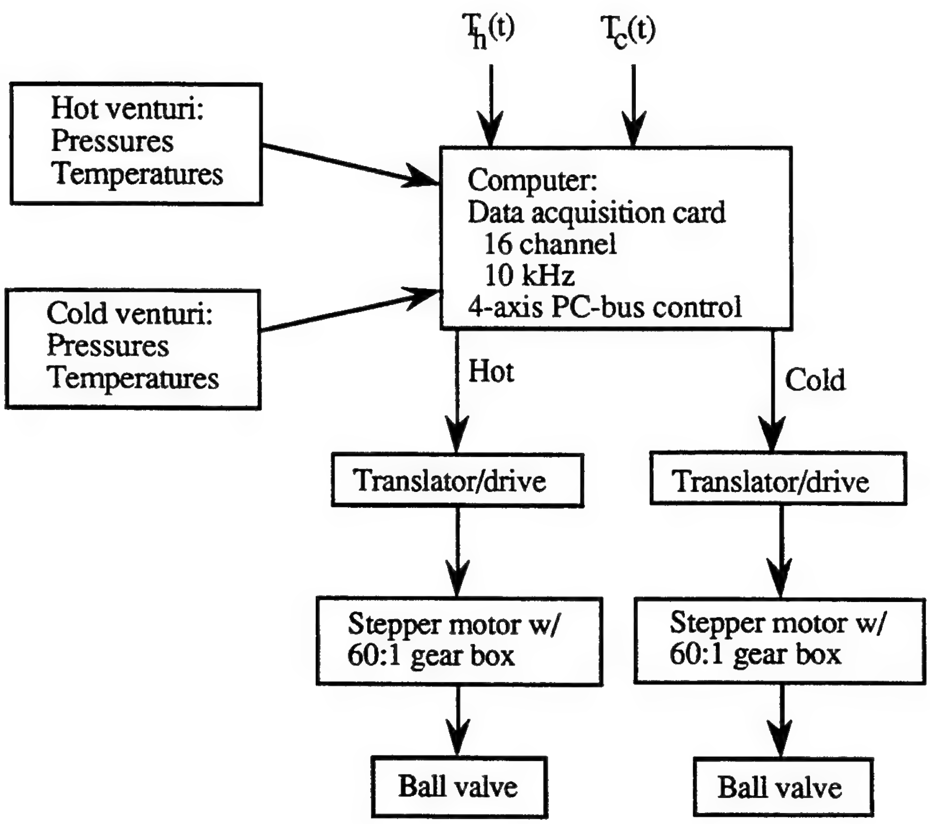

Figure 3 - Schematic of venturi control system.

2.1.2.2 Heat Exchanger Analysis

Performance of the heat exchanger is analyzed in this section. Consider a gas flow through a tube as shown in Figure 4.

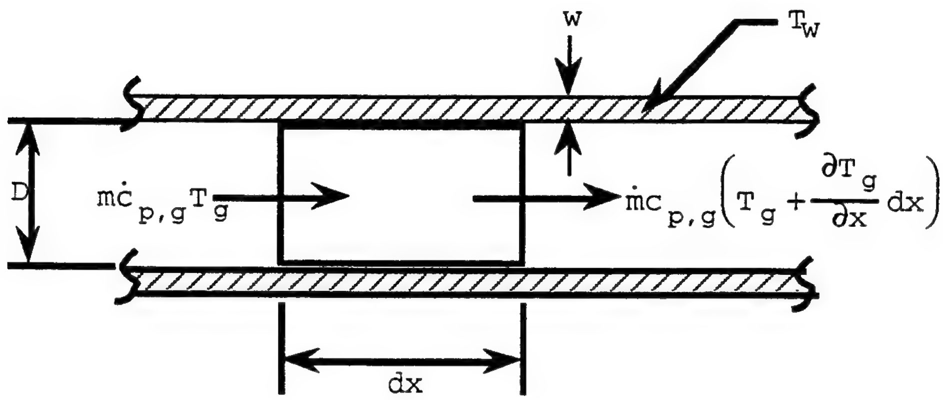

Figure 4 - Differential flow element within a tube.

An energy balance on the above differential fluid element yields

$$dq = \dot{m}c_{p,g}\frac{\partial T_g}{\partial x}dx = h\pi Ddx\left(T_w - T_g\right) \tag{11}$$

or

$$\frac{\partial T_g(x,t)}{\partial x} = \frac{h\pi D}{\dot{m}c_{p,g}}\left[T_w(x,t) - T_g(x,t)\right] \tag{12}$$

A similar balance on a differential wall element yields

$$dq = \rho(\pi Dwdx)c_{p,w}\frac{\partial T_w}{\partial t} = -h\pi Ddx\left(T_w - T_g\right) \tag{13}$$

or

$$\frac{\partial T_w(x,t)}{\partial t} = \frac{-h}{w\rho c_{p,w}}\left[T_w(x,t) - T_g(x,t)\right]. \tag{14}$$

Nondimensionalizing the temperature in equations (9) and (11) on

$$\theta_g = \frac{T_g - T_{g,i}}{T_{w,i} - T_{g,i}} \tag{15}$$

and

$$\theta_w = \frac{T_w - T_{g,i}}{T_{w,i} - T_{g,i}} \tag{16}$$

yields

$$\frac{\partial \theta_g}{\partial x} = \frac{h\pi D}{\dot{m} c_{p,g}} \left(\theta_w - \theta_g \right) \tag{17}$$

and

$$\frac{\partial \theta_w}{\partial t} = \frac{-h}{w \rho c_{p,w}} \left(\theta_w - \theta_g \right). \tag{18}$$

By inspection one can determine two constants given by

$$\lambda = \frac{\dot{m} c_{p,g}}{h\pi D} \tag{19}$$

and

$$\tau = \frac{w \rho c_{p,w}}{h} \tag{20}$$

on which to nondimensionalize the length (x) and time (t), respectively. The constant λ represents the tube length required for the gas to reach 1/e of the tube wall temperature if the tube wall temperature were constant. Similarly, the constant τ represents the time required for the tube wall to reach 1/e of the gas temperature if the gas temperature were constant. Nondimensionalizing then yields

$$\frac{\partial \theta_g}{\partial x^*} = \left(\theta_w - \theta_g \right) \tag{21}$$

and

$$\frac{\partial \theta_w}{\partial t^*} = -\left(\theta_w - \theta_g \right) \tag{22}$$

where $x^*=x/\lambda$ and $t^*=t/\tau$.

The above set of coupled non-linear differential equations may be numerically integrated to obtain a solution as a function of nondimensional time and distance. The solution is shown in Figure 5.

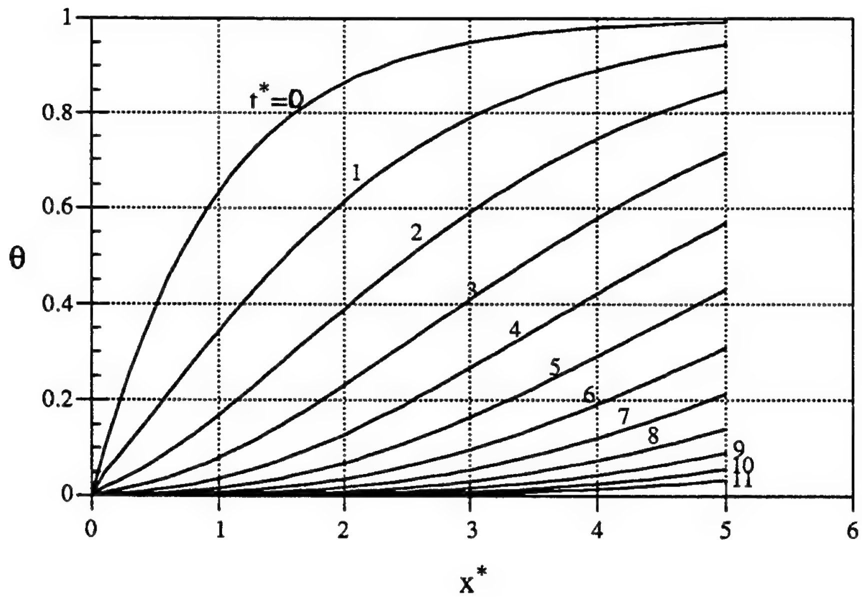

Figure 5 - Heat exchanger performance.

Various heat exchanger configurations can be evaluated by plotting their performance on the above plot. The criteria for deciding whether or not a particular heat exchanger configuration was acceptable was determined using three conditions: 1). The heat exchanger should be long enough so that the exit gas temperature is at the desired level without excessively pre-cooling the heat exchanger. It was felt that cooling the heat exchanger to a temperature 10 °F cooler than the exit gas flow would be acceptable. For typical inlet and exit gas temperatures of 70 °F and -100 °F, respectively, this translates into a nondimensional exit temperature of θ=0.94. 2). The tubes should be sufficiently thick that the increase in exit temperature (which corresponds to a decrease in θ) should not be excessive. It was felt that a decrease in θ of 5% would be tolerable. 3). The cost of the tubes for the heat exchanger should be limited to less than $3000.

Table 1 presents a summary of various heat exchanger configurations that were analyzed. It can be seen that although all of the above satisfy the cost condition (condition

3), only configurations 9 and 11 satisfy the first two conditions. Configuration 11 is the preferred choice as it has the smaller temperature decay (from 0.98 to 0.95) and has the added advantage of being the shorter of the two. With configuration 11, the temperature is expected to increase about 5 deg. F during the test duration. No significant advantage from a technical point of view was found by using steel or copper tubes instead of aluminum.

	t_w (in)	OD (in)	L (ft)	N	x*	t*	θ_i	θ_f	wt. (lbs)	($)
8	0.049	0.5	10	222	2.71	0.231	0.93	0.86	133	1465
9	0.028	0.25	7	956	4.53	0.454	0.99	0.94	154	2744
10	0.028	0.25	6	956	3.88	0.454	0.98	0.90	132	2352
11	0.035	0.25	6	1111	4.26	0.369	0.98	0.95	185	2733

Table 1 - Summary of heat exchanger performance characteristics for Al tube, N_2 at 7.7 lb/s.

2.1.2.3 Control Valve Sizing

In this section, the control valves are sized based on the design cooling system condition. The nitrogen gas in the coolant tank reservoir is assumed to be at 70 °F and 200 psia. A technique will have to be devised to charge this reservoir from the tuber farm. The decay of the reservoir pressure and temperature are inconsequential as long as there is an adequate supply. The existing six, 200 psia tanks will be more than adequate for this purpose. The initial conditions in the cooling system are assumed to be given as

Reservoir: $T_r(0)=70$ °F

$P_r(0)=200$ psia

Vane cooling flow: $T_h(0)=T_r(0)=70$ °F

$T_c(0)=-136$ °F

$T_m(0)=-70$ °F

$m_m(0)=11.5$ lb/s

Blade cooling flow: $T_h(0)=T_r(0)=70$ °F

$T_c(0)=-136$ °F

$T_m(0)=-95$ °F

$m_m(0)=6.5$ lb/s

The mass flow rates of the hot and cold gas streams necessary to produce a coolant gas stream at the desired temperature and mass flow rate at any instant in time can be calculated from the above information using Equation 1 through 3 and Equation 6. The corresponding control valve area needed can then be calculated from Equation 8.

The variation in control valve area as a function of time for the above conditions over the expected test duration is shown in Figures 6 and 7. The variation can be approximated using a linear fit for both the vane and blade cooling flows. Different control valve areas can be obtained by changing either the pintle or diffuser geometry.

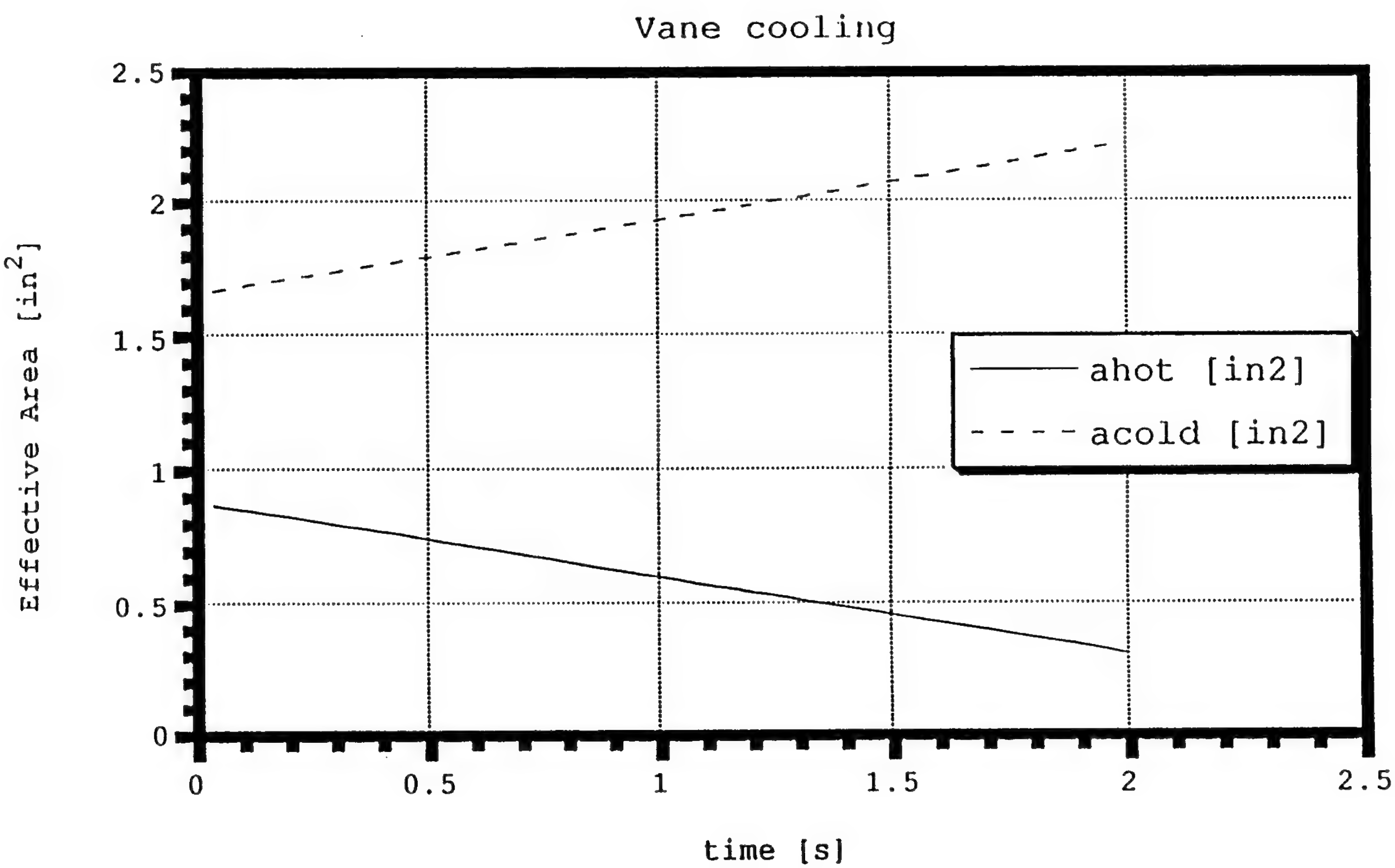

Figure 6 - Variation in control valve area for vane cooling

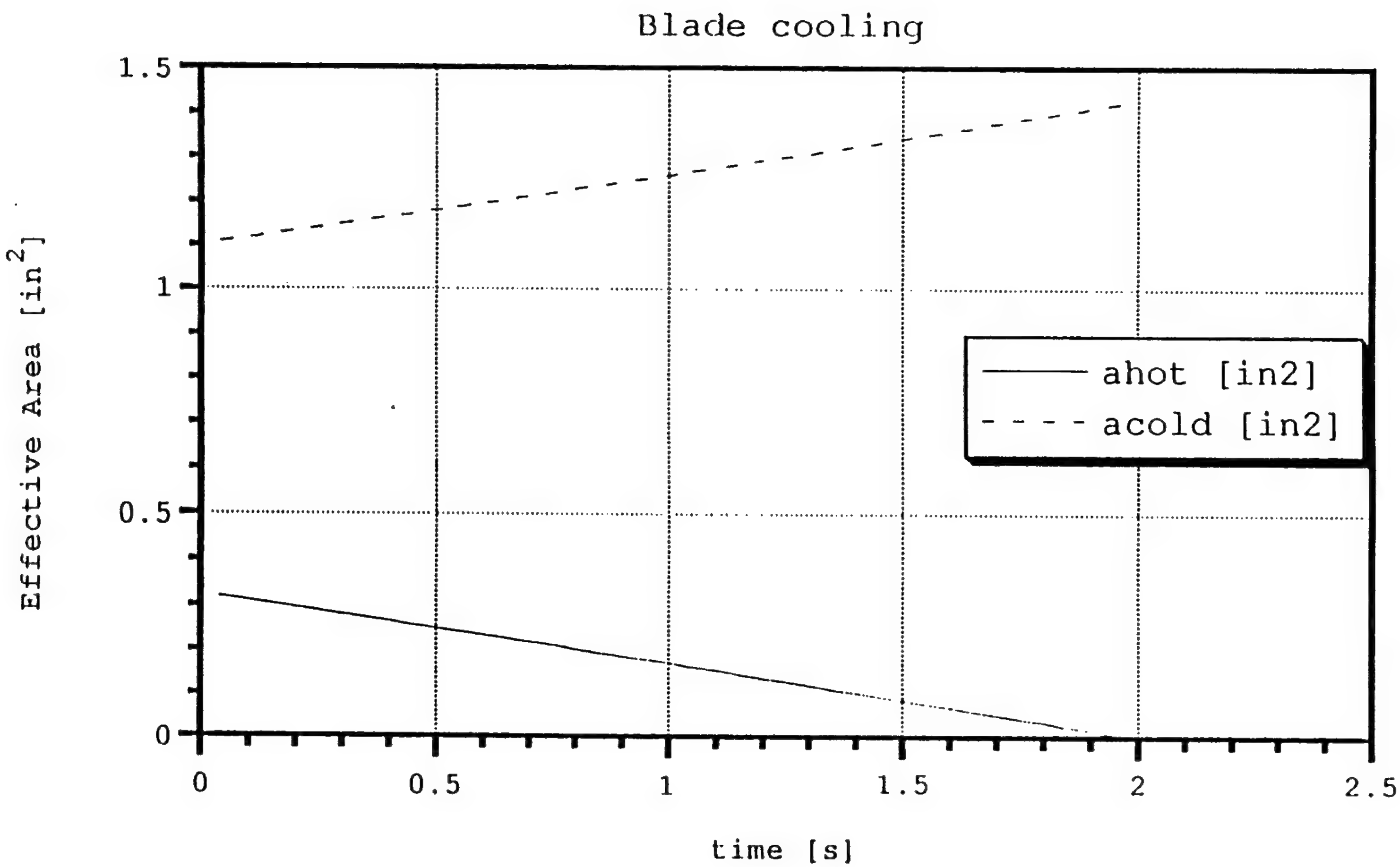

Figure 7 - Variation in control valve area for blade cooling

2.1.2.4 Heat exchanger pre-cooling system analysis

The heat exchanger is cooled to the desired temperature using a mix of liquid nitrogen (LN_2) and N_2 gas. The liquid nitrogen is stored in a dewar under pressure from a nitrogen source which pumps the LN_2 to an atomizer. N_2 gas at room temperature is metered and throttled, and then used to vaporize the liquid nitrogen to produce a gas stream at the desired temperature. This gas stream is passed through the heat exchanger and vented to atmosphere.

The heat exchanger analysis described in the previous section was used to determine the mass flow rates of LN_2 and N_2 gas needed to cool the heat exchanger to the required temperature in a reasonable time. Specifically, the question posed was "For an incoming N_2 gas stream at -100 °F, what is the flow rate needed to produce a heat exchanger (configuration 11 in Table 4) exit flow of -95 °F within 30 minutes?" Calculations similar to those performed in the heat exchanger analysis suggested that a flow rate of 0.22 lb/s would be adequate to cool the heat exchanger to the desired temperature in the required time.

The mass flow rates of LN_2 and air can be calculated from the conservation of mass and energy equations to be $\dot{m}_{LN_2}$=0.0646 lb/s and $\dot{m}_{N_2}$=0.156 lb/s. For 30 minutes of flow, 65 liters of LN_2 are required. A 200 liter LN_2 dewar should provide more than adequate cooling capacity.

2.1.3 - Control System Instrumentation

It is important that the instrumentation chosen for use in the cooling facility (pressure transducers and thermocouples) be tailored for use in short-duration facilities. The relatively high frequency response of some available pressure transducers enables them to be used without a problem in this facility. The thermocouples, however, must be sized

such that the error due to their thermal lag (the transient error) is smaller than the maximum error in temperature that one is willing to accept. The thermocouple rakes to be used in the cooling system are similar to those in use at Calspan and those to be used to measure the temperature of the core gas flow and should be sufficiently responsive for this application.

2.1.4 - Control System Development Issues

There are several areas of the control system design which have not yet been finalized and present significant technical challenges. The first area is the dynamic control of the venturi valves. This sub-system is the most complex of all the components in the cooling system. It is the only sub-system with parts which are required to move during the run. Furthermore, the computer must interact with the MCS system regarding coolant flow initiation. The pintle position needs to be positioned quickly and accurately. Currently, it is envisioned that the computer will have sufficient computational capability to calculate the venturi position in a small fraction of the test time. The pintle will move to a specified position, dwell there a specified length of time, them move to another position when the signal from the computer is received. If the above scheme proves to be inadequate, it may be necessary to incorporate a "predictive" scheme which accounts for the computation time, and moves the valves at a specified speed during this time. This should be relatively easy to incorporate.

The second issue is that of measuring the actual cooling gas mass flow rate being delivered to the turbine stage. The Fox venturi metering valves are not only used to control the flow but also to measure the flow rate. To measure the flow rate accurately, the pintle position of the valves must be known very accurately. This will require a calibration of the entire control system to ensure that the pintle is accurately positioned and recorded during the test period. It is questionable whether any control system will be stable enough to meet the stringent accuracy goals of ATARR.

With the issues described above in mind, a decision was made to revisit the simple blowdown cooling system that was investigated during the early stages of this program. The following section describes the blowdown system.

2.2 - Blowdown Cooling System

The second system considered for delivery of cooling gas to the turbine stage housed within the ATARR test section is more simple in concept than the control system previously discussed. Rather than actively controlling the mass flow and temperature of the cooling gas, the blowdown system approximates the core flow function by following the same isentropic blowdown physics.

2.2.1 - Blowdown System Theory

The desired core flow to cooling gas ratios can be written in terms of the isentropic blowdown Equations 1 through 3.

$$\frac{T_{cf}(t)}{T_c(t)} = \frac{T_{cf}(0)\left[1+\frac{t}{\tau}\right]^{-2}}{T\ (0)\left[1+\frac{t}{\tau}\right]^{-2}} = \text{CONSTANT} \tag{23}$$

$$\frac{P_{cf}(t)}{P_c(t)} = \frac{P_{cf}(0)\left[1+\frac{t}{\tau}\right]^{\frac{-2\gamma_{cf}}{\gamma_{cf}-1}}}{P\ (0)\left[1+\frac{t}{\tau}\right]^{\frac{-2\gamma_c}{\gamma-1}}} = \frac{P_{cf}(0)}{P_c(0)}\left[1+\frac{t}{\tau}\right]^{\frac{-2(\gamma_c-\gamma_{cf})}{(\gamma_c-1)(\gamma_{cf}-1)}} \tag{24}$$

$$\frac{\dot{m}_{cf}(t)}{\dot{m}_c(t)} = \frac{\dot{m}_{cf}(0)\left[1+\frac{t}{\tau}\right]^{\frac{-1-\gamma_{cf}}{\gamma_{cf}-1}}}{\dot{m}\ (0)\left[1+\frac{t}{\tau}\right]^{\frac{-1-\gamma_c}{\gamma-1}}} = \frac{\dot{m}_{cf}(0)}{\dot{m}_c(0)}\left[1+\frac{t}{\tau}\right]^{\frac{-2(\gamma_c-\gamma_{cf})}{(\gamma_c-1)(\gamma_{cf}-1)}} \tag{25}$$

These equations assume that the cooling system blowdown process is isentropic which may not be entirely true because of the unavoidable heat transfer and losses within the system's pipes and valves. However, this difficulty is alleviated to some extent by initiating the coolant gas flow for a brief period of time prior to opening the main valve in order to precool the plumbing. These equations also assume that the freestream blowdown time constant τ is perfectly matched to the cooling system blowdown time constant. Under these conditions the temperature ratio is a true constant while the pressure and mass flow ratios are still functions of time because of the difference in core flow and cooling gas specific heat ratios.

The difference between specific heat ratios is unavoidable because of the large temperature difference required between the core flow and cooling gases. The influence of this difference in Equations 24 and 25 was investigated at the following typical conditions:

Cooling gas specific heat ratio,	$\gamma_c = 1.40$
Freestream gas specific heat ratio,	$\gamma_{cf} = 1.30$
Blowdown time constant,	$\tau = 38$ s
Test duration,	$t = 2$ s

The results of this analysis, shown in Figure 8, indicate that the mass flow and pressure of the cooling gas don't decay as quickly as they do in the core flow. For the conditions given, the ratios drop by 8% by the end of the test period. The initial cooling system pressure and mass flow could be reduced by 4% to generate the correct flow conditions at mid-test. Therefore, there would be a 4% deviation in mass flow and pressure during the test.

The cooling system blowdown time constant, given by Equation 26, must match the core flow time constant.

$$\frac{1}{\tau} = \dot{m}_{corr}\frac{(\gamma_c-1)\sqrt{\gamma_c R T_c(0)}(A_1+A_2)}{2\gamma_c V} \tag{26}$$

where
$$\dot{m}_{corr} = \gamma\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} \tag{27}$$

The initial temperature within the cooling system blowdown tank T_c is set by the core flow supply tank initial temperature and the desired temperature ratio. This in turn sets the specific heat ratio of the nitrogen, γ_c. The tank volume V is incrementally adjustable through the use of multiple cooling system supply tanks. The two choke areas (A_1+A_2) are adjusted to produce the desired mass flow rate and time constant. The test section supply line choke area A_1 sets the mass flow rate to the blades or vanes and the bleed-off choke area A_2 sets the additional area necessary to obtain the desired time constant.

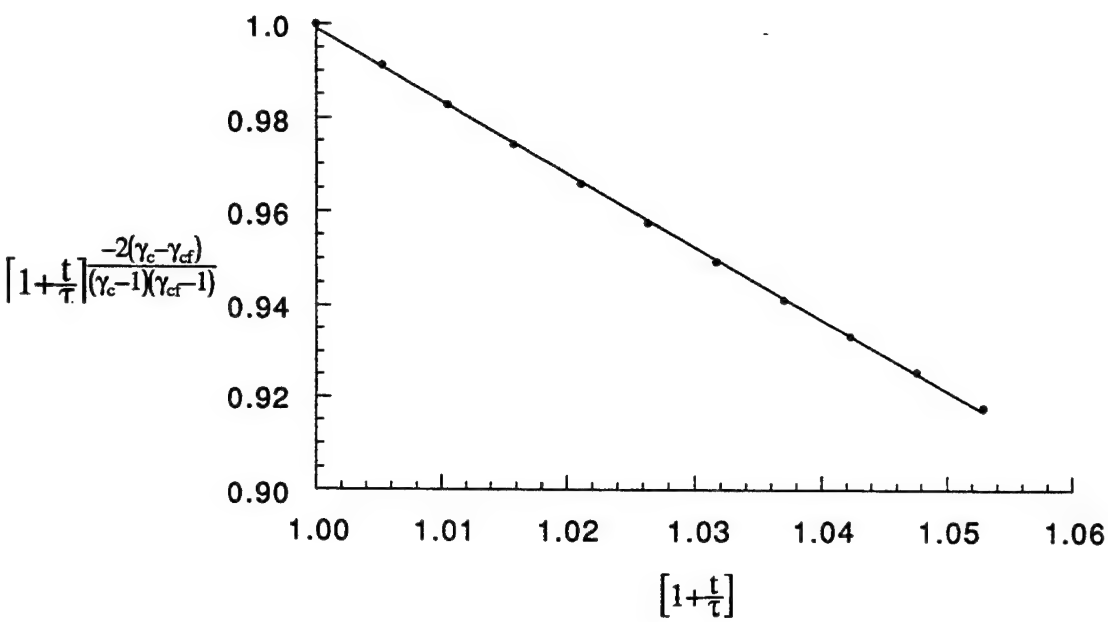

Figure 8 - Variation of mass flow and pressure ratios due to specific heat ratio difference between flows.

2.2.2 - Blowdown System Operation

A schematic of the proposed blowdown cooling system is given in Figure 9. The test procedure for the system begins by releasing approximately 35 liters of liquid nitrogen into each of the blowdown supply tanks for their initial cool down. This cool down process should require approximately 30 minutes not including the time required to dispense the LN2 into the tank. The analysis followed in estimating the amount of LN2 and time required for the cool down is given in Section 2.2.3. Most runs will require two of the existing 500 gallon tanks, which are to be insulated, but some runs may require as many as three.

Once the supply tanks are cooled to the required gas temperature the tank pressure should be approximately 130 psia. Depending on the run, this may be slightly greater than or less than the required initial pressure. If the tank pressure is greater than required, the tank may be vented using valve V7. Further adjustment to the temperature may be performed by adding N2 or LN2 with valves V4 and V3, respectively, and venting the tank as necessary. If the tank pressure is low, the appropriate amounts of LN2 and N2 may be added to the tank separately to avoid a cryogenic explosion within the manifold.

Once the initial supply tank pressure and temperature is established, the area of the Fox venturi valves is set by adjusting the pintle of each and locking them into place. The initial supply tank pressure and temperature and the main cooling line and bleed-off choke areas are determined by a model of the system which will be discussed in Section 2.2.4. The test is initiated by opening the fast acting valve, V6, with the MCS system, and terminated by closing valve V6. The fast acting valve, V6 would be designed with a fail safe position of being closed.

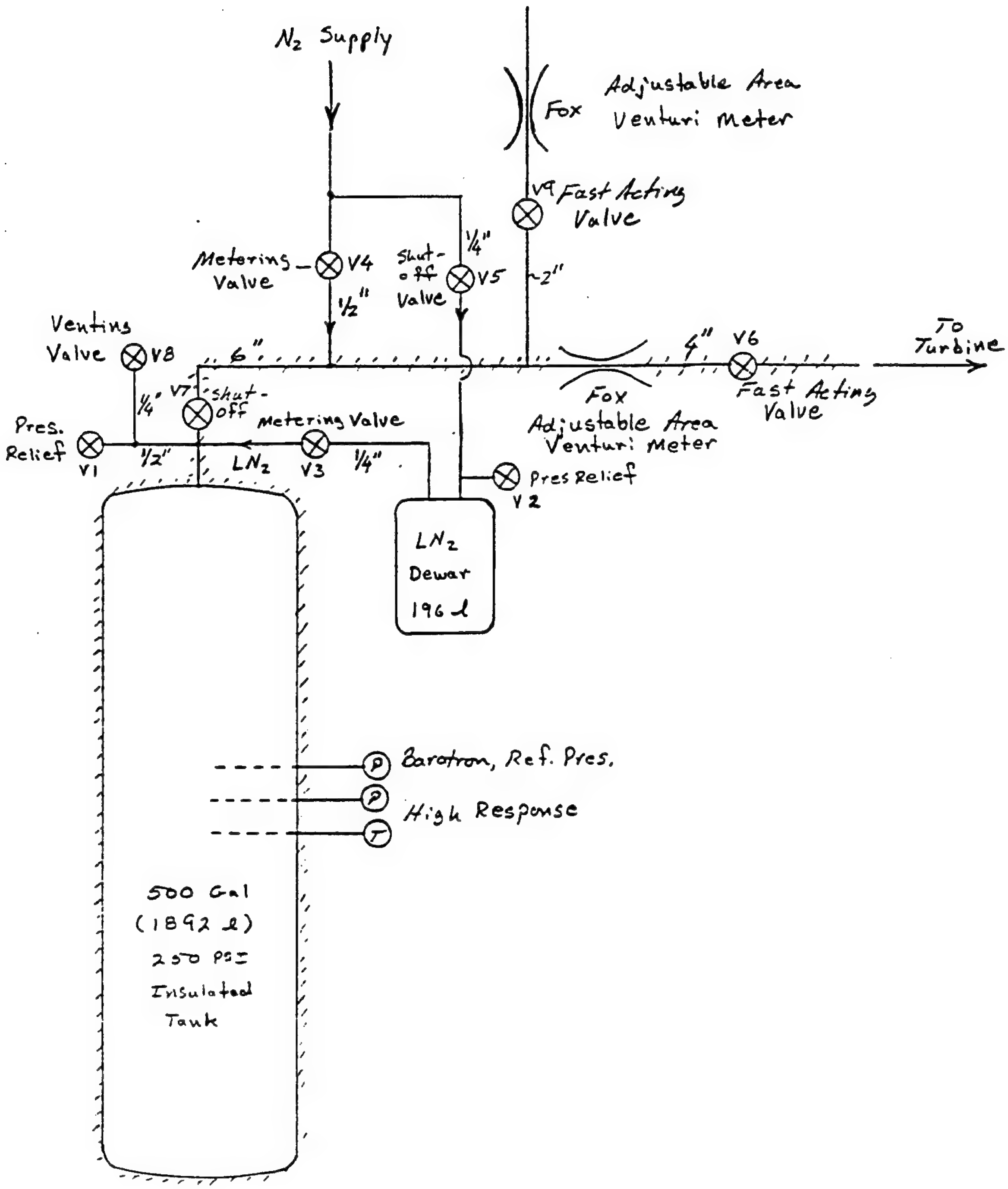

Figure 9 - Blowdown Cooling System Schematic

2.2.3 - Blowdown System Analysis

2.2.3.1 LN2 Volume Required for Tank Cool Down

The existing 500 gallon, 250 psia tanks have the following dimensions: 3' 1" dia., 9' 10" tall, 1/4" wall, stainless steel. The mass of one tank is approximately 1000 lbm or 453.6 Kg. This represents a significant mass that must be cooled to the desired temperature without using an unreasonable amount of LN2. Consider the differential energy balance equations for the liquid nitrogen and metal given by Equations 28 and 29, respectively.

$$dQ = dm_{LN2}\left[h(T) - h_{sat\,liq}\right] = dm_{LN2}\left[\left(T - T_{sat\,liq}\right)C_{p,N2} + h_{fg,LN2}\right] \tag{28}$$

$$dQ = -m_m C_{p,m} dT \tag{29}$$

Where h in the above equations represents enthalpy.

Equating 28 and 29 and solving the resulting differential equation yields an expression for the mass of LN2.

$$m_{LN2} = -m_m \frac{C_{p,m}}{C_{p,LN2}} \ln\left[\frac{\left(T - T_{sat\,liq}\right)C_{p,N2} + h_{fg,LN2}}{\left(T_{i,m} - T_{sat\,liq}\right)C_{p,N2} + h_{fg,LN2}}\right] \tag{30}$$

The process occurs between atmospheric pressure and 130 psia. To estimate the mass of LN2 required, Equation 30 was evaluated at both the initial and final pressures and the average value of the LN2 mass was found to be 28.8 Kg which equals 34.5 liters. This is a reasonable amount of LN2 and can be provided by the 196 liter dewar as planned.

2.2.3.2 Time Required to Cool Tank

Another important test parameter is the time required for the tank to come into thermal equilibrium. The controlling factor is the low heat transfer coefficient between the

gas and the tank walls since this is the main thermal resistance. Consider the case where the mass of the tank is exposed through a heat transfer coefficient, h, to nitrogen gas at temperature $T_g(t)$. For simplicity, it is assumed that the LN2 has completely vaporized. This will yield a conservative estimate. The differential amount of heat transferred between the gas and the metal is given by

$$dQ = hA[T_m(t) - T_g(t)]dt \tag{31}$$

The differential relation between the amount of temperature increase in the gas and the temperature decrease in the metal is given by

$$dQ = m_g C_{p,g} dT_g = -m_m C_{p,m} dT_m \tag{32}$$

Integrating Equation 32 and solving for $T_g(t)$ yields

$$T_g(t) = -K[T_m(t) - T_m(0)] + T_g(0) \tag{33}$$

where

$$K = \frac{m_m C_{p,m}}{m_g C_{p,g}} \tag{34}$$

Substituting Equation 33 into 31 and integrating yields

$$\frac{T_m(t)(1+K) - KT_{m,i} - T_{g,i}}{T_{m,i}(1+K) - KT_{m,i} - T_{g,i}} = e^{-\frac{t}{\tau}} \tag{35}$$

where

$$\frac{1}{\tau} = \frac{hA}{m_m C_{p,m}}(1+K) \tag{36}$$

The time constant τ represents the time required for the metal temperature to decay by $\left(1-\frac{1}{e}\right)$ of the initial metal to gas temperature difference. Assuming the conditions given below,

$$h = 5\ \frac{W}{m^2K} \text{ (typical of natural convection)}$$

$$A = 8.85\ m^2 \text{ (inside surface area of tank)}$$

the time constant is determined to be 10 minutes, which is quite reasonable. Thus, the time requiredto achieve 95% of the total cool-down temperature would be 30 minutes.

2.2.4 - Blowdown System Model

The basis of a computer model for the blowdown system has been written. There are three main objectives for this model. The first objective is for the model to predict the flow through the system as a function of time in order to determine how well the blowdown system will match the turbine core flow properties and thus produce the desired ratios. The losses and heat transfer in the system should be modeled to some degree to ascertain their effects on the assumed isentropic blowdown process. The second objective is for the model to serve as a design tool. It will be required to finalize the system configuration and size the components. The third objective is for the code to become part of the ATARR facility model. In this capacity, it would be used to predict the initial supply tank pressure and temperature and the required main cooling line and bleed-off choke areas. The code requires further development and refinement to meet the model objectives, but it does operate and can be useful as will be described below.

2.2.4.1 Basic Equations and Integration

The current blowdown cooling system model is an unsteady compressible code which makes the following assumptions:

1.) isentropic flow

2.) adiabatic flow

3.) inviscid flow

4.) Nitrogen at the conditions considered is an ideal gas.

5.) One dimensional flow into and out of each chamber.

6.) Change in potential and kinetic energy within the chambers is negligible.

The unsteady form of the conservation of mass used in the current blowdown cooling system model is given by Equation 37.

$$\frac{dm}{dt} = \Sigma\, \dot{m}_{in} - \Sigma\, \dot{m}_{out} \tag{37}$$

The simplified form of the first law of thermodynamics applied in the model is given by Equation 38.

$$\frac{dE}{dt} = \Sigma\, \dot{m}_{in} h_{in} - \Sigma\, \dot{m}_{out} h_{out} \tag{38}$$

The equations of state used in this model are those of a thermally and calorically perfect gas and are given by Equations 39 and 40, respectively.

$$P = \rho RT \tag{39}$$

$$h = C_p T \tag{40}$$

The method used to integrate Equations 37 and 38 is formulated below using the chamber mass, Equation 37, as an example. This is the same method used in the main ATARR system model.

$$\int_{m(t)}^{m(t+\Delta t)} dm = \int_{t}^{t+\Delta t} (\Sigma\dot{m}_{in} - \Sigma\dot{m}_{out})dt \tag{41}$$

The mass flow rates within the integral are assumed constant over the time interval Δt and removed from the integral.

$$m(t+\Delta t) = m(t) + (\Sigma\dot{m}_{in} - \Sigma\dot{m}_{out})\Delta t \tag{42}$$

The mass flow rates are evaluated at the initial and final conditions of each time step and an average of the two values is used in updating of the chamber properties. The initial mass flow rate is calculated using the chamber properties from the previous time step and the appropriate gas dynamic equation for flow with area change. The chamber properties are temporarily updated based on this initial mass flow rate and then used to calculate a final mass flow rate.

2.2.4.2 Stability of System Model

The code is not inherently stable because the technique of modeling the chambers as reservoirs is not ideal for this particular flow system. Many of the chambers have a relatively low volume to flow area ratio and can't be well modeled as a reservoir. Several schemes were employed to help stabilize the model by preventing or correcting for excessive flow into or out of a particular chamber. The first level check ensures that not all the mass or energy within a chamber has been expelled. If the net mass or energy flow calculated will result in a negative chamber mass or energy, it is limited to 80% of what was within the chamber. A second check requires the flow within a chamber to be subsonic and reduces the mass flow rates in an iterative fashion as necessary.

The next level of checks prevent an unrealistic flow rate by examining the final chamber pressures from a fluid mechanics point of view. The updated total pressure within a chamber is limited to an estimated minimum final pressure that could occur if the current chamber and downstream chamber went to an equilibrium state due to flow between the two. The flow out of the chamber is reduced, if necessary, in an incremental and iterative fashion until this condition is met. The updated static pressure within the chamber is also checked and limited to a maximum value equal to that of the upstream chamber's total pressure. Again the flow rate is reduced as necessary in an incremental and iterative manor. Care is taken within the code to update the appropriate chamber properties when any of the flow rates are changed.

2.2.4.3 Routine for Setting System Initial Conditions

The system model estimates the initial supply tank pressure and temperature and main cooling line and bleed-off choke areas at the beginning of each run of the code. It is known that the mass flow rate and pressure ratios will not be matched throughout the test

period. Therefore, the model attempts to match all the desired conditions at a point in the middle of the test period to minimize any deviations.

The model requires as input, the mid-test point temperature and pressure within the cooled airfoil chambers as well as the mass flow out of the airfoils. These input parameters can be determined from the known turbine inlet conditions at a point in the middle of the test period, and the desired core flow to cooling gas ratios. The airfoil chamber pressure is determined from the surface pressure distribution of the airfoil at or near the mid-test point conditions, the film cooling hole configuration and the cooling gas mass flow rate. Likewise, the airfoil chamber temperature can be determined from the desired core flow to cooling gas temperature ratio.

The model assumes the flow to be steady at this mid-test point and marches upstream from the airfoil chamber through to the supply tank. The chamber properties and Mach numbers are calculated from the given geometry and mass flow rate using gas dynamic equations under the given model assumptions. The main line choke area is calculated in the process and is based upon choked flow through the valve at the given mass flow rate.

The initial cooling supply tank pressure and temperature are calculated from the mid-test point tank conditions using Equations 2 and 3 respectively. The bleed-off choke area is calculated from Equations 26 and 27 with the main line choke area A_1 already determined. These conditions are then used in the unsteady blowdown analysis which follows.

2.2.4.4 Current Status of Model

As mentioned previously, the model is not fully operable and doesn't meet the ambitious goals outlined above. The model does run in limited cases where the cooling

system configuration is simple and the turbine inlet pressure is controlled. The results of one such test case are included in this section to illustrate the status of the model.

A schematic of the cooling system analyzed as a test case is given in Figure 10, where the chambers are numbered 1-5. Chamber 1 is the cooling system supply tank(s), chamber 4 is the turbine test section and chamber 5 is the bleed-off line. This simplified configuration has all the essential elements: supply tanks, a main line venturi choke valve, bleed-off line with a choke valve, and fast acting valves for activation of both the main and bleed-off lines.

The input file used in the test case is given in the Appendix along with a hard copy of the FORTRAN code. For the test case three of the 500 gallon tanks were required to supply the volume needed to obtain the desired blowdown time constant of 43.5 s. The turbine inlet pressure was started at 1 psia and stepped up to 30 psia after 40 ms. This low pressure impulse was analyzed out of necessity because the model can not yet contend with a realistic impulse. A pressure differential of 3.5 psi was specified between chambers 3 and 4 as a mid-test point target. This specified pressure differential was to produce a 5 lbm/s cooling gas mass flow rate at the mid-test point (1s).

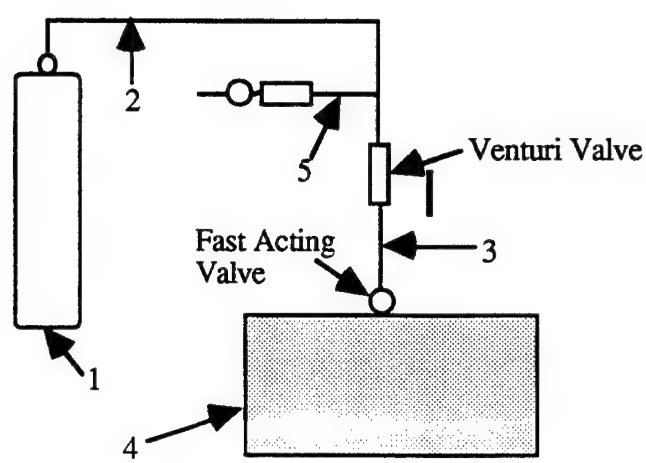

Figure 10 - Cooling system model test case schematic

The results of the test case run are given in Figures 11 through 17. Figure 11 gives the Mach number within each chamber as a function of time. Figure 12 shows how the chamber pressures decay with time. A large pressure drop exists across the venturi choke between chambers 2 and 3. The pressure within chamber 3 does not decay at the blowdown rate because it is on the downstream side of the system's choke. The pressure differential between chambers 3 and 4 at the mid-test point is greater than the target value of 3.5 by 1.9 psi. The blowdown time constant was determined from Equation 43 using the initial and final supply tank pressures. It was found to be approximately 61 s which is significantly larger than the target value of 43.5 s.

$$\tau = \frac{t}{\left[\left(\frac{P(t)}{P(0)}\right)^{\frac{(\gamma-1)}{-2\gamma}} - 1\right]} \tag{43}$$

Figure 13 gives the chamber total temperatures which appear to be equal and constant throughout the system. Under the adiabatic and isentropic assumptions with no external work the total temperature should be constant for a steady system. However, the total temperature in this transient blowdown system should decay according to Equation 3. The total temperature is calculated using the caloric equation of state for a perfect gas, Equation 44, with the updated enthalpy and mass from the integration. Therefore, the temperature is not explicitly held constant or forced to conform to Equation 3.

$$T = \frac{H}{mC_p} \tag{44}$$

Figure 14 gives the Mach number at the exit area of each chamber and shows that the main line venturi valve becomes unchoked shortly after the mid-test point. Both the venturi valves should be sized so that they remain choked to produce the correct blowdown time constant and mass flow rate. The main line choke area was determined using a factor of 0.9 times the critical area calculated in the mid-test point sizing routine to help ensure that

the valve remained choked. This was obviously not a small enough factor and should be decreased as necessary.

Figure 15 shows the mass flow rates from each chamber and how they decay with time. The bleed-off flow rate remains nearly constant while the other flow rates decay at the blowdown rate. Figure 16 gives the mass in chambers 2,3 and 5 as a function of time. Figure 17 gives the supply tank mass as a function of time and shows that only a portion of the overall mass is expelled.

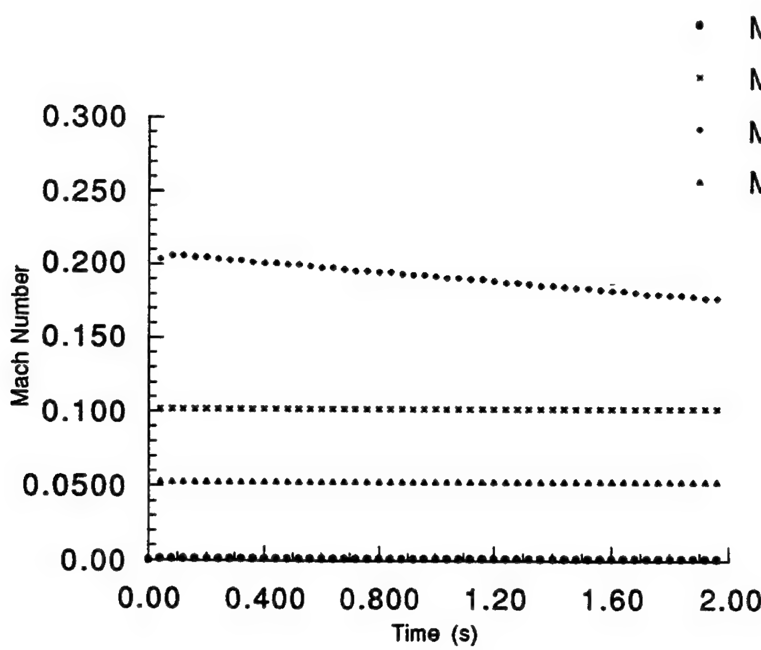

Figure 11 - Chamber Mach Number

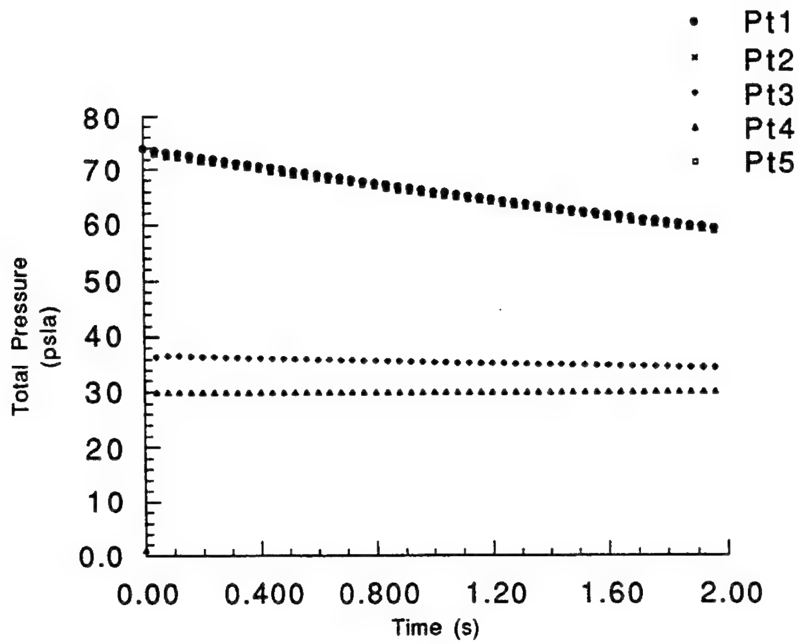

Figure 12 - Chamber Total Pressure

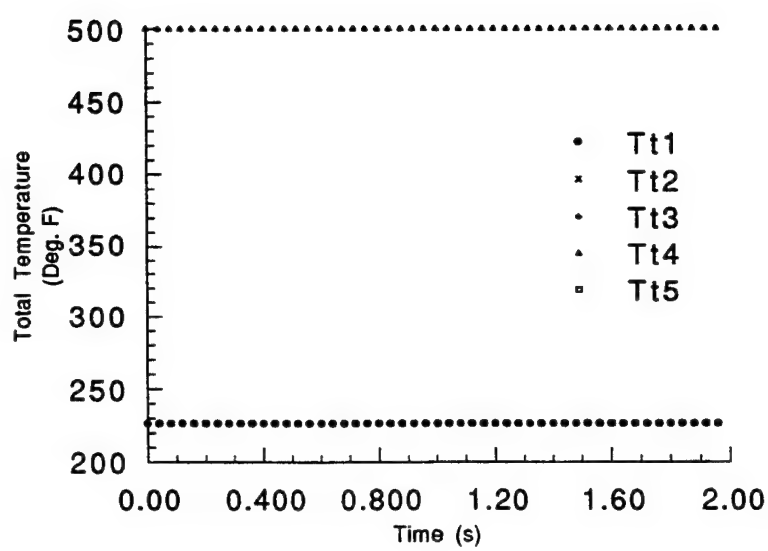

Figure 13 - Chamber Total Temperature

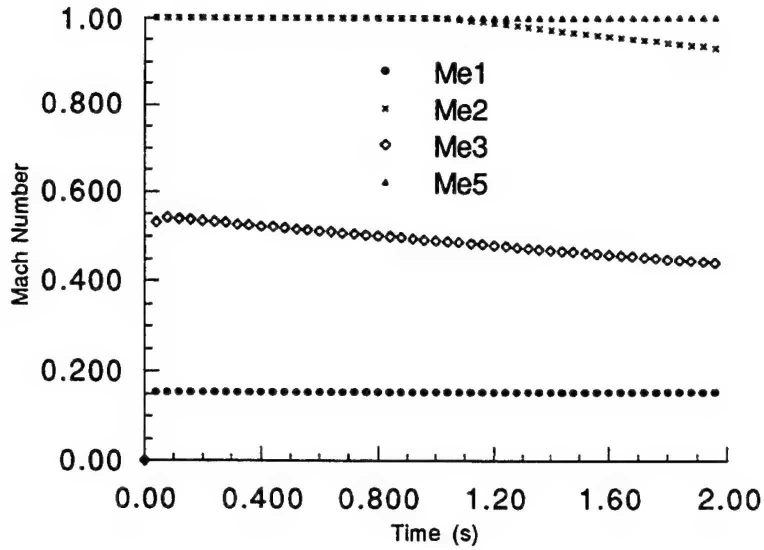

Figure 14 - Mach Number at the Chamber Exit

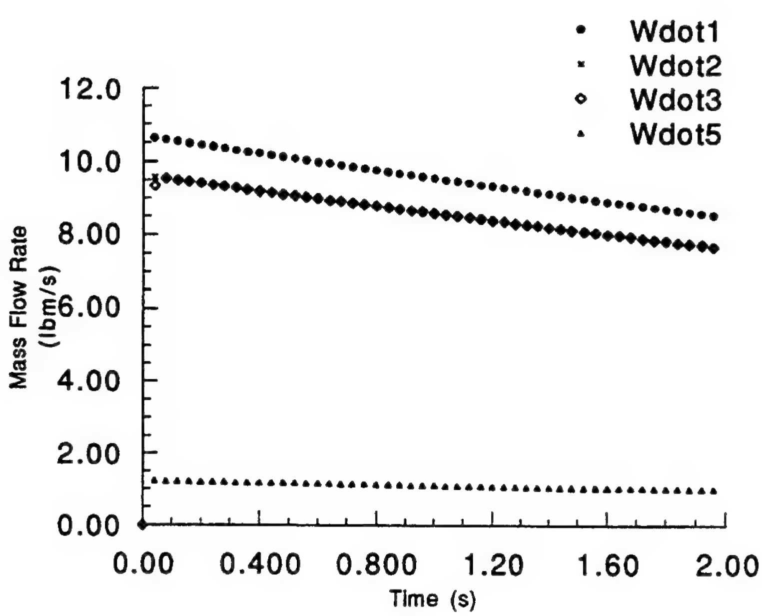

Figure 15 - Chamber Exit Mass Flow Rate

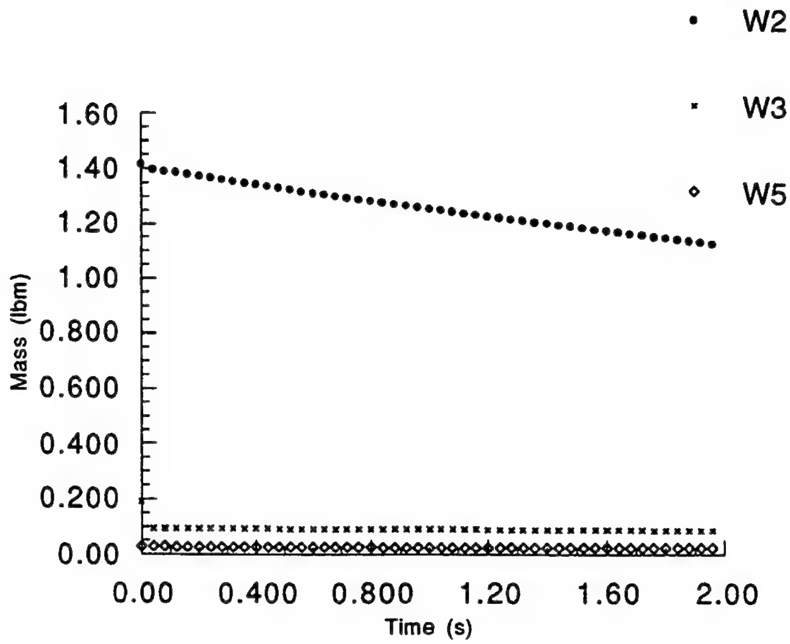

Figure 16 - Chamber Mass

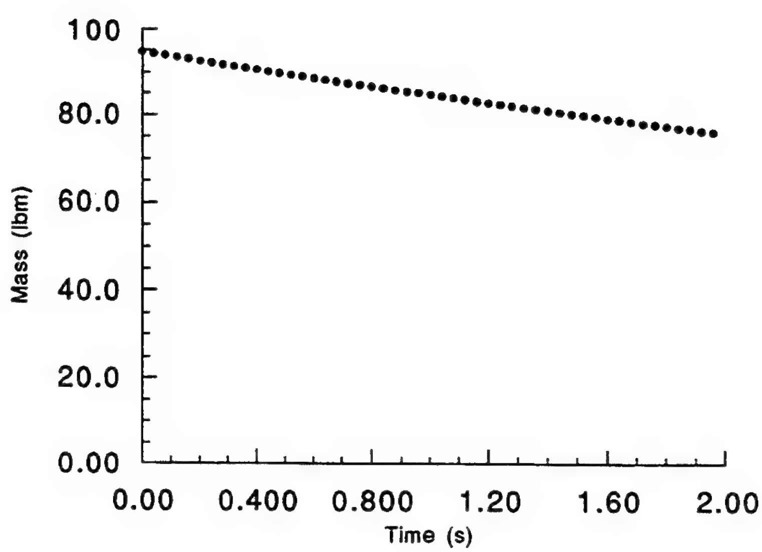

Figure 17 - Supply Tank Mass

The results indicate that the model does conform to the conservation of mass and the first law of thermodynamics. However, there is a question whether the model predicts the transient behavior of the system. Stability of the model is still an issue and must be addressed further for the model to be able to analyze more realistic and complex configurations. Eventually, the heat transfer and pipe losses should be modeled to determine their effects which could be significant.

2.2.5 - Blowdown System Instrumentation

The same instrumentation issues brought up by the control cooling system in Section 2.1.3 apply to the blowdown system with one additional concern. The supply tank pressure in the blowdown system must be accurately measured during the transient process to determine the blowdown time constant and mass flow rates. During the blowdown process the transducer will be subjected to a thermal transient which can significantly affect the transducer characteristics. Therefore, a high response pressure transducer capable of withstanding cryogenic conditions must be calibrated as a function of temperature as well as pressure. This type of pressure and temperature calibration is under development for both ATARR and AHWT programs..

2.2.6 -Remaining Blowdown System Development Issues

The most significant unresolved issue associated with the blowdown cooling system is to complete the development of an unsteady flow model for predicting the behavior of the system. This model must be capable of predicting the time-dependent blowdown process and of predicting the necessary initial tank conditions and required choke areas. It is felt that the model is the only obstacle remaining to be resolved before completing the blowdown system design.

3.0 - CONCLUSIONS AND RECOMMENDATIONS

This last chapter of the report will attempt to objectively compare and contrast the two systems with reference to the four main system requirements outlined in Section 1.1. Unfortunately, not all the comparisons have clear conclusions at this time and additional information may be required to make a definitive decision on which cooling system will best suit the needs of the ATARR facility.

3.1 - Matching Freestream to Cooling Gas Ratios

It appears at this point that the control cooling system would be more able to deliver the desired cooling gas properties and flow rate required to match the time dependent core-gas flow parameters. However, the feedback control system required to control the metering valves hasn't been demonstrated, and the blowdown system model has not answered all of the questions concerning the blowdown systems capabilities. Therefore, more information would be helpful in making a final decision.

3.2 - Accuracy of Flow Measurements

The actual cooling gas flow delivered to the vanes and blades could be best measured by the blowdown system. This is based on the fact that it will be very difficult to accurately know the history of the venturi pintle positions over the test period with the control system. It is essential that the valve positions be known very accurately in order to make accurate mass flow measurements. Calibration of the feedback control and valve

system could pose a significant obstacle in the development of the control system. On the other hand, the blowdown system is not without its problems. The temperature compensation of the supply tank pressure transducers will be a difficult task.

3.3 - Development Costs

A detailed costing of the two systems may be misleading at this stage in their respective designs; therefore, a comparison of the large items unique to each system is presented instead.

Control System Unique Hardware

Description	Source	Qty.	Cost $
Valve Control System	Various	2	5788
Heat Exchanger	Calspan	1	5452
Atomizer	BEX	1	100
Cryogenic Shut-off Valve, 4" (V8)	Jamesbury	1	1500
Cryogenic Shut-off Valve, 1" (V9)	Jamesbury	1	300
High Resp. Pres. Transducer	Endevco	1	900
High Resp. Thermocouple, Type K	Omega	3	750
Thermocouple, Type K, 1/8" sheath	Omega	1	52
		Total	14,842

Blowdown System Unique Hardware

Description	Source	Qty.	Cost $
Insulation of Existing 500 gal Tanks	R.A. Kramig	3	13,950
Fast Acting Valve, 2" (V9)	Flowdyne	1	12,000
Cryogenic Vent Valve, 1/4" (V8)	Hoke	1	198
		Total	26,148

It appears on the basis of material costs that the control system would be approximately $11,000 less expensive than the blowdown system. However, this difference in equipment costs is estimated to be only 5% of the total projected cooling system cost ($250,000) and could be easily spent in unforeseen engineering development efforts. The engineering development time is more difficult to determine and will be the real deciding factor in the cost of the two systems. At present, the blowdown system appears to require less engineering development because similar systems have been used at Calspan previously. Although some effort is required to finish the blowdown model, the control system development issues are more difficult.

3.4 - Ease of Operation and Maintenance

Both systems, once in place, will be relatively easy to operate and maintain. Each system requires a cool down process, the supply tank(s) in the blowdown system and the heat exchanger in the control system. Both systems require preliminary analysis to determine what the cooling gas flow should be as a function of time to match the known core flow conditions. The blowdown system will require some additional effort to set the initial supply tank pressure and temperature which are not critical for the control system. Finally, both system are activated with fast acting valves through the MCS system. The

maintenance of the control system would be slightly greater than the blowdown system because of the moving parts in the control valve system. The two system are very comparable from an operation and maintenance viewpoint.

3.5 - Recommendations

It was not possible to demonstrate completely that the blowdown-type cooling system can meet all of the system requirements as currently envisioned. However, we believe that this is the best cooling system with which to initiate the ATARR operation. We feel that this system will be able to match the desired core flow to cooling gas ratios of interest for an acceptable portion of the test duration and to the degree of accuracy required.

Much of the hardware between the two systems is common. Procurement of this common hardware could be initiated at little risk. A blowdown system could be built from this common hardware using a mylar diaphragm instead of a fast acting valve for the bleed-off line. The system could be run in an experimental mode with inexpensive insulation on one of the existing 500 gallon tanks. This experimental system would give some definitive answers to questions about a blowdown system. The experimental system could then be developed into either a complete blowdown system or a control type of system.